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THEORETICAL MODELING AND EXPERIMENTAL STUDIES ON VEGETABLE OIL FUELED CONVENTIONAL ENGINE

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Abstract- Experimental analysis of the engine with various vegetable oil and its blends requires enormous effort, money and time. Hence a theoretical method based on various existing process models has been developed for the performance evaluation of a compression ignition engine by using diesel and Jatropha straight vegetable oil as an input fuel. The performance tests are carried out on a compression ignition engine using diesel, jatropha oil (unheated) and jatropha oil (preheated) as fuel. The effects of fuel injection pressure and fuel inlet temperature on the engine performance and emission for different fuels are also analyzed using this model. The comparisons of theoretical and experimental results are discussed. Engine performance characteristics predicted by this model are in closer approximation to that of experimental results.
Key words- Diesel engine, Jatropha oil, Engine simulation, Engineering equation solver.

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Introduction

Diesel engines are widely used as power sources in medium and heavy duty applications because of their lower fuel consumption and lower emissions of carbon monoxide and unburned hydrocarbons compared with gasoline engines. Rudolf Diesel ran an engine on groundnut oil at the Paris Exposition of 1900.Since then, vegetable oils have been used as fuels when petroleum supplies were expensive or difficult to obtain. With the increased availability of petroleum in the 1940s, research into vegetable oils decreased. Since the oil crisis of the 1970s research interest has expanded in the area of alternative fuels [1].The difficulties associated with using raw vegetable oils in diesel engines identified in the literature are injector coking, severe engine deposits, filter gumming problems, piston ring sticking, injector coking and thickening of the lubricating oil. The high viscosity and low volatility of raw vegetable oils are generally considered to be the major drawbacks for their utilization as fuels in diesel engines [2].However; these effects can be reduced or eliminated through preheating, blending, ultrasonically assisted methanol transesterification and supercritical methanol transesterification. [3, 4].The rapid development of computer technology has techniques to find out the effect of the fundamental processes in the engine systems.

The theoretical models used in case of internal combustion engines are mainly described as thermodynamic and fluid dynamic models. Models based on thermodynamics can be further classified as single zone heat release model, phenomenological jet based model and multi-zone model. Single zone models assume that the cylinder content is uniform in composition and temperature and are suitable for prediction of engine performance. Phenomenological combustion models are based on each individual processes occurring in engine cycle such as fuel injection, mixture formation, heat release, heat transfer and emission formation. Multi-zone models incorporate the development of the fuel spray with time and simplified quasi steady equations are used to describe processes like fuel injection, atomization, air entrainment, droplet formation, evaporation, wall impingement, ignition, heat release, heat transfer and so on. Fluid dynamic based models, often called multi-dimensional or computational fluid dynamics (CFD) models are based on solving the governing equations for conservation of mass, momentum and energy and species con-

centration through a definite discretization procedure [5].
In LHR engine, during the tests it was observed that more fuel coming from injectors heated up the fuel in the fuel tank. This reduced the fuel viscosity, so the need for preheating was eliminated [6].Various methods of using Jatropha oil and methanol such as blending, transesterification and dual fuel operation were studied experimentally. They found brake thermal efficiency was better in the dual fuel operation and with the methyl ester of Jatropha oil as compared to the blend [7].A correlation were developed for predicting the ignition delay of two biodiesels in a direct-injection diesel engine. Biodiesels from palm and rapeseed oils show shorter ignition delay than diesel fuel [8]. The new proposed correlations for biodiesels were compared against the fossil diesel. The comparison with ignition delay measurements show that the new correlations predict ignition delay for biofuels better than the available correlations for diesel fuel [9]. Computational fluid dynamics (CFD) models are extensively used for flow visualization,
fuel-air mixing combustion analysis and turbulent diffusion combustion studies by different researchers [10].Research has been carried out in the field of fuel injection, injection rate, injection duration and injection pressure on engine performance and emissions during the 1990s.

Theoretical modeling of compression ignition engines

A mono zone thermodynamic model is used for analyzing the performance characteristics of four stroke indirect injection compression ignition engines. A spatial uniformity of pressure, temperature and composition in each combustion chamber, at each instant of time, is assumed. The gas medium is assumed to obey the perfect gas law. The fuels considered are diesel and jatropha oil. The molecular formula of diesel is approximated as $C_{12}H_{22}$. Based on the properties, the molecular formula of Jatropha oil is approximated as $C_{18}H_{32}O_2$ [11,12]. Polynomial expressions are used for each species (O_2, N_2, CO_2, H_2O) considered in the calculation of specific heats, internal energy and enthalpy as a function of temperature [5] [13].

Energy conservation

By applying the first law of thermodynamics for main and pre combustion chamber as open system, the net heat release rate is determined as follows[5]:

$$\frac{dQ_{n,1}}{dt} - p_1\frac{dV_1}{dt} + h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_1}{dt} \tag{1}$$

$$\frac{dQ_{n,2}}{dt} - p_2\frac{dV_2}{dt} - h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_2}{dt} \tag{2}$$

Where the volume of the pre-chamber V2 is constant so, Eq. (2) becomes

$$\frac{dQ_{n,2}}{dt} - h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_2}{dt} \tag{2a}$$

Where U1and U2 are gas sensible internal energies, h2,1 as the gas sensible specific enthalpies,m2,1 is the mass flow between the two chambers , Qn,1 and Qn,2 are net heat releases which are the difference between the combustion energy release (gross) and the heat transferred (loss) to the chamber walls.

The perfect gas state equations for the two chambers are as:
$p_1V_1=m_1RT_1$ $\quad$ $p_2V_2=m_2RT_2$
Therefore, the corresponding gross heat release rates, which are the energies released from the combustion of fuel, are given by:

$$\frac{dQ_{g,1}}{dt} = \frac{dQ_{n,1}}{dt} + \frac{dQ_{w,1}}{dt} \tag{3}$$

$$\frac{dQ_{g,2}}{dt} = \frac{dQ_{n,2}}{dt} + \frac{dQ_{w,2}}{dt} \tag{4}$$

Where $\left(\frac{dQ_{w,1}}{dt}\right)$ and $\left(\frac{dQ_{w,2}}{dt}\right)$ are the rates of heat transferred to the walls of the main chamber and the pre-chamber.

By knowing the lower heating value Θ of the fuel, the fuel burned mass rate in each chamber is computed from:

$$\frac{dm_{b,1}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,1}}{dt} \tag{5}$$

$$\frac{dm_{b,2}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,2}}{dt} \tag{6}$$

Considering the each combustion chamber as open system, gas mass balance in differential form gives:

$$\frac{dm_1}{dt} = \frac{dm_{2,1}}{dt} + \frac{dm_{f,1}}{dt} \tag{7}$$

$$\frac{dm_2}{dt} = -\frac{dm_{2,1}}{dt} + \frac{dm_{f,2}}{dt} \tag{8}$$

Where m1 and m2 are the gas mass in each chamber consisting of the corresponding air plus fuel, mf,1 and mf,2 are the fuel mass existing in each chamber. The existing fuel mass in each chamber is the quantity due to combustion and the fuel transported with the mass flow rate (dm/dt) exchanged through the connecting throat. This is expressed mathematically as:

$$\frac{dm_{f,1}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,1}}{dt} + \frac{m_{f2,1}}{m_{2,1}}\frac{dm_{2,1}}{dt} \tag{9}$$

$$\frac{dm_{f,2}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,2}}{dt} - \frac{m_{f2,1}}{m_{2,1}}\frac{dm_{2,1}}{dt} \tag{10}$$

Where, if dm2,1/dt>0 then mf2,1 /m2,1=mf,2/m2,and if dm2,1/dt<0 then mf2,1/m2,1=mf,1/m1.
It is obvious, that all the previous Eq. from (1) to (10) are in the form of differentiations with respect to time t .These can be converted to differentiations with respect to degrees crank angle θ ,by multiplying by the factor dt/dθ = 1/6N.Where N is the engine speed.

Volume at any Crank Angle

The combustion chamber volume (main + pre- chamber) at any crank angle position can be computed using the following equation:

$$V(\theta)=V_d\left[\frac{r}{r-1}-\frac{1-\cos\theta}{2}+\frac{1}{2}\sqrt{\left(2\frac{L}{S}\right)^2-\sin^2\theta}\right] \quad (11)$$

Where r is the compression ratio, L is the length of connecting rod, S is the stroke length, Vd is the displacement volume and θ is the angular displacement with respect to BDC.

Combustion reaction and properties calculation

Considering the fundamental stoichiometric chemical equation for the hydrocarbon -oxygen complete combustion can be explained as [14]:

$C_nH_m+[n+(m/4)]O_2 \rightarrow nCO_2+(m/2)H_2O$ (12)

Where n is the no. of moles of carbon and m is the number of moles of hydrogen.

Internal energy (U), enthalpy (H), specific heats at constant pressure (Cp) and constant volume (Cv) of the gaseous mixture are calculated on the basis of charge composition and temperature. These properties of gaseous mixture are calculated as follows [5].

U (T) =A+ (B-R)*T+Cln (T) (13)

H (T) =A+B*T+C*lnT (14)

C_p (T) =B+C/T (15)

C_v (T) = (B-R) +C/T (16)

Where A, B and C are the coefficients of the polynomial equation.

Heat release analysis

Gross heat release due to combustion is calculated by using single Weibe, s heat release correlation [5

$$\frac{dQ_g}{d\theta}=a(m+1)\left(\frac{Q_r}{\Delta\theta}\right)\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^m \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \quad (17)$$

Where Qr is the heat released per cycle.θ0 is the start of combustion, Δθ is the combustion duration The value of m for all the fuel is taken as 2.0 and value of a as taken 5.0.

Heat transfer analysis

The heat transfer between each chamber trapped mass and surrounding walls is calculated by using the following formula (Annand, 1963):

$$\frac{dQ_w/d\theta}{A}=a\frac{k}{D}(\mathrm{Re})^b\left(T_g-T_w\right)+c\left(T_g^4-T_w^4\right) \quad (18)$$

Re=ρ D u_p /μ

u_p = 2NS/60

Where A is the heat transfer area, Tw is the temperature of the combustion chamber walls, Tg is the gas temperature, Re is the Reynolds number, k is the thermal conductivity , D is the cylinder bore. up is the mean piston speed, S is the piston stroke, ρ is the gas density and μ is the dynamic viscosity.

For both the combustion chamber constant b =0.75 and c =0 for compression period and otherwise c=3.3x 10-8 [14] a=0.30 for main chamber [15].

Mass of fuel injected

Considering that nozzle open area is constant during the injection period, total mass of the fuel injected for each crank angle is calculated as follows:

$$m_f=C_dA_n\sqrt{2\rho_f\Delta P}\left(\frac{\Delta\theta_f}{360N}\right)n \quad (19)$$

Where n is the number of injector nozzle holes, C_d is the coefficient of discharge of injector nozzle, A_n is the cross sectional area of nozzle, ΔP is the pressure drop in the nozzle, N is the engine speed and $\Delta\theta_f$ is the fuel injection period.

Pressure drop in the nozzle

The pressure drop in the nozzle is calculated as follows [16]:

$$\Delta P=0.5\rho_f\left(\frac{u_{inj}}{C_d}\right)^2 \quad (20)$$

Where u_{inj} the spray velocity from nozzle hole is given as:

$$u_{inj}=\left(\frac{dm_f}{d\theta}\right)\left(\frac{6N}{\rho_fA_n}\right) \quad (21)$$

Where $\left(\frac{dm_f}{d\theta}\right)$ is the fuel injection rate (kg/^{0}CA)

$$\left(\frac{dm_f}{d\theta}\right)=\left(\frac{m_f}{n\Delta\theta_f}\right) \quad (22)$$

Soot concentration

Soot is a fine dispersion of black carbon particles in a vapor carrier. The main source of soot is from the incomplete hydrocarbon combustion. Soot particles form, grow, and oxidize as a result of chemical reactions that occur during combustion. The detailed of soot modeling are mentioned by [17]. Exhaust gas soot concentration related to normal conditions are as follows:

$$[C]_e=\int_{\theta}^{480}\frac{d[C]}{dt}\frac{d\theta}{6\eta}\left(\frac{0.1}{p}\right)^{\frac{1}{\gamma}} \quad (23)$$

Where γ =1.33 is an exhaust gas adiabatic exponent, p is the cylinder pressure, [C] is the current soot concentration in the cylinder and θ is the crank angle.

Theoretical solution procedure

The equations of the model adopted in the previous section are suitable for any hydrocarbon fuel as diesel, vegetable oil and bio-diesel etc. These equations are solved numerically using with time step size of 2^0 crank angles. The engine geometrical parameters, molecular weight of gaseous products and various constants used in the modeling are defined. The input parameters used in modeling are injection pressure, crank angle and the molecular formula of the diesel and jatropha oil. The several physical, chemical and thermal properties are also measured and calculated as input parameters as shown in Table 1.

Table 1- Instruments to measure various properties and their values

Property	Instrument	Measured and calculated value of Diesel	Measured and calculated value of Jatropha oil
Density (kg/m^3)	Hydrometer	817	910
Kinematic viscosity at 70^{0}C (cSt)	Rotational viscometer	3.5	12
Surface tension at 27^{0}C (N/m)	Multi frequency ultrasonic interferometer	0.028	0.042
Calorific value(MJ/kg)	Bomb calorimeter	43.04	37.08

The properties of gaseous constituents such as enthalpy, internal energy and specific heats are calculated as a function of temperature. The pressure and temperature of the gases in the combustion chamber are calculated for every two degree crank angle. From the typical design of the engine, the combustion chamber volume at every degree crank angle is calculated with the help of (Eq.11). The outputs of the modeling program are instantaneous pressure, temperature, volume and the performance and emission parameters that includes brake thermal efficiency, brake specific fuel consumption and soot density. The performance and emission characteristics of the engine fueled with jatropha oil (preheated and unheated) and diesel are analysed by this model .The engine model is analysed for the variation in fuel injection pressure for different fuels. An engineering equation solver program, developed by F-Chart Software is used for simulation.

In brief, the numerical solution stages consists of calculation of compression phase, calculations of combustion and expansion phases and calculation of the mean state of zones.

Effect of density of air on heat transfer per crank angle

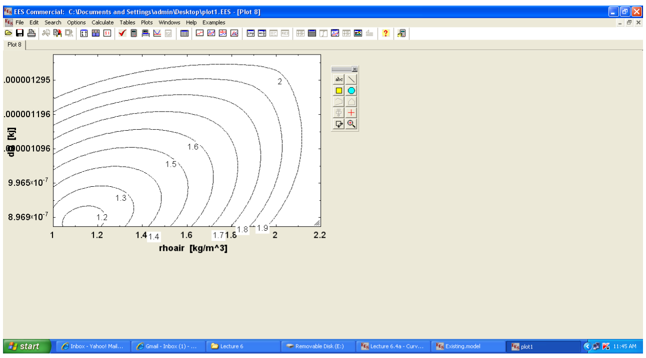

Fig. 3.1- Variation of heat transfer per crank angle vs density of air

The variations of heat transfer per crank angle with respect to density of air shown in the Figure 3.1.This variation is shown in the form of contour. As density of air increases in the combustion chamber, heat transfer is also increases. So such types of variations of heat transfer and air density verify the programming in the software.

Net work done out put with crank angle

The net work done with crank angle variations is shown in Figure 3.2.The negative work done in the figure shows that the work is done on the system in the compression phase, which is from 225^0 to 345^0.After the fuel injection in the combustion chamber burning take place. At this stage pressure rise in the cylinder take place and expansion phase start which shows the positive work done or in other word we can say that the work done by the system. So same manner as above this variation also verify the out put of the program.

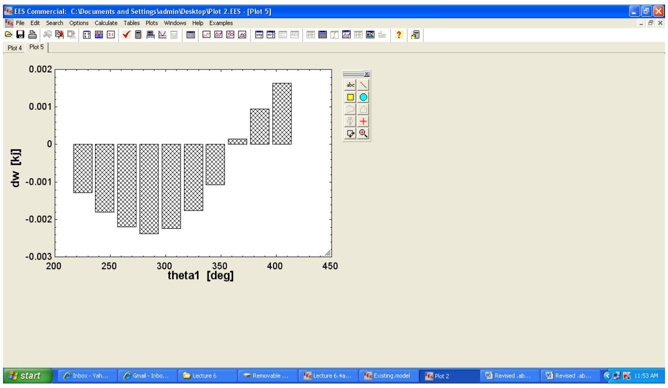

Fig. 3.2- Variation of work done vs crank angle

Experimental set up

In the present investigation, Jatropha oil, a non -edible type vegetable oil, is chosen as a potential alternative fuel and it is used as the fuel in compression ignition engines. The oil content of Jatropha seed ranges from 30% to 40% by weight and the kernel itself ranges from 45% to 60 %. Fresh Jatropha oil is slow-drying, odorless and colorless oil, but it turned yellow after aging. Some of the properties of Jatropha oil fall within a fairly narrow band and are quite close to those of the diesel oil. The main problem of using Jatropha oil in unmodified form in diesel engine is its high viscosity. Therefore, it is necessary to reduce the fuel viscosity before injecting it in the engine. High viscosity of Jatropha oil can be reduced by heating the oil using waste heat of exhaust gases from the engine. Various procedures followed and the instruments used are given in Table 1. Viscosity of Jatropha oil and diesel was measured at different temperatures to find the effect of temperature on viscosity.

The engine selected for present study is widely used, mostly for agricultural irrigation purpose and also in many other small and medium scale commercial applications like producing electricity, running flour mills, rice mills. A single cylinder, four stroke, vertical, water cooled, indirect injection diesel engine was selected for the experiments. The technical specifications of the engines are given in Table 2.

Table 2- Engine specifications

Manufacturer	Field Marshal Engine Ltd.,India
Model	FM-4
Engine type	Vertical,4-stroke,single cylinder,indirect-injection,water cooled, compression ignition engine
Rated power	7.35 kW at 1000 rpm
Bore/stroke	120/139.7 (mm)
Compression ratio	17
Nozzle	DL30S12002MICO
Nozzle holder	9430031264MICO
Fuel pump plunger	9x03/323MICO
Nozzle opening pressure	145 bar
Sump capacity	4.5 litre

The engine can be started by hand cranking using decompression lever. The test engine is coupled with a single phase, 230V AC alternator of 7.5 kVA capacity to absorb maximum power produced by the engine. The schematic layout of the experimental setup for the present investigation is shown in Figure 10.

The main components of the experimental setup are two fuel tanks (Diesel and Jatropha oil), fuel conditioning system, heat exchanger, exhaust gas line, by-pass line, and performance and emissions measurement equipment. Two fuel filters are provided next to the Jatropha oil tank so that when one filter gets clogged, supply of fuel can be switched over to another filter while the clogged filter can be cleaned without stopping the engine operation. The engine is started with diesel and once the engine warms up, it is switched over to Jatropha oil. After concluding the tests with Jatropha oil, the engine is again switched back to diesel before stopping the engine until the Jatropha oil is purged from the fuel line, injection pump and injector in order to prevent deposits and cold starting problems. This purging typically takes about 15 min at idling. A counter flow shell and tube type heat exchanger is used to preheat the vegetable oil using waste heat of the exhaust gases. In order to control the temperature of the Jatropha oil within a range of 75-100⁰C, a by-pass valve was provided in the exhaust gas line before the heat exchanger. A thermocouple was provided in the exhaust line to measure the temperature of the exhaust gases. Exhaust gas opacity was measured using smoke meter (Make: AVL Austria, Model: 437).

Experimental test analysis

Experiments were conducted for optimizing fuel injection pressure for Jatropha oil and diesel. Finally, performance and emission tests were conducted for diesel, unheated Jatropha oil, preheated Jatropha oil. These tests were conducted in three phases. In first phase, tests were conducted by diesel, while changing the fuel injection pressure. These tests were conducted with diesel to generate baseline data. In the second phase, tests were conducted using unheated Jatropha oil , while operating the engine on varying fuel injection pressure .In the third phase, tests were conducted using preheated Jatropha oil while operating the engine on varying fuel injection pressure . The performance and emission data were then analyzed for all experiments.

Results and discussion

The experimental measurements covered the spectrum of loads from 1.84, 3.68, 5.52 , 7.35 and 9.0 kW .In this work , the result are presented for the speed of 1000 rpm at corresponding loads, during the closed part of the cycle.

Higher viscosity is a major problem in using vegetable oil as fuel for diesel engines. In the present investigations, viscosity was reduced by heating. Viscosity of Jatropha oil was measured at different temperatures in the range of 30-100⁰C. The results are shown in Figure 1.

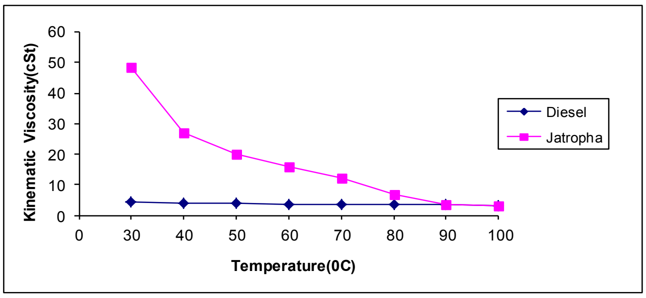

Fig. 1- Variation of viscosity with temperature

Viscosity of Jatropha oil decreases remarkably with increasing temperature and it becomes close to diesel at temperature above 90⁰C . Viscosity of diesel was 3.8 cSt at 40⁰C. For Jatropha oil, viscosity was found 3.16 cSt at a temperature 100⁰C. Therefore, Jatropha oil should be heated to 100⁰C before injecting it into the engine in order to bring its physical properties close to diesel (at 40⁰C).

Effect of varying fuel injection pressure on performance and emission of engine

Brake thermal efficiency, Brake specific fuel consumption and smoke opacity were measured and calculated at different fuel injection pressures for diesel as well as pre heated Jatropha oil. For diesel fueled engine, BSFC decreases as the fuel injection pressure increases from 175 bars to 195 bars Figure3.

Further increase in fuel injection pressure results in increased BSFC. BTE was found to increase with increasing fuel injection pressure from 175 bars to 195 bars Figure 2. However, increase in fuel injection pressure from 195 bars to 205 bars showed decrease in thermal efficiency. Maximum thermal efficiency (30.8%) was found at fuel injection pressure of 195 bars. It can be seen from Figure 4 that increase in fuel injection pressure from 175 bars to 195 bars resulted in decreased smoke opacity.

However, further increase in fuel injection pressure from 195 bars to 205 bars showed increased smoke opacity. Therefore, smoke opacity was lowest at a fuel injection pressure of 195 bars. Based on BSFC, BTE and smoke opacity, 195 bar was found optimum fuel injection pressure for diesel.

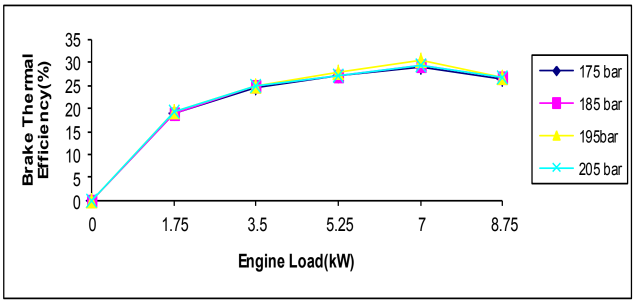

Fig. 2- Effect of fuel injection pressure on brake thermal efficiency of diesel fueled engine

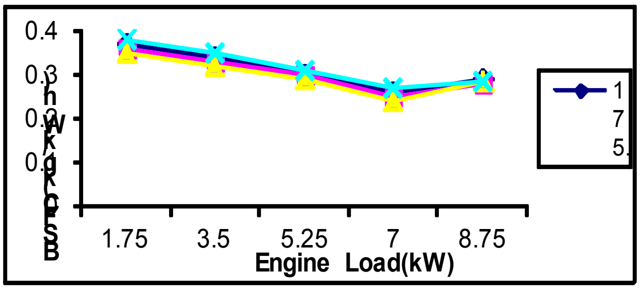

Fig. 3- Effect of fuel injection pressure on brake specific fuel consumption of diesel fueled engine

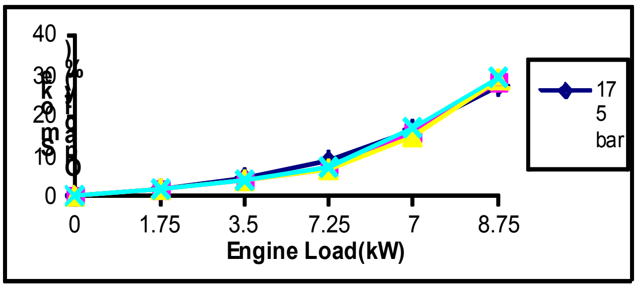

Fig. 4- Effect of fuel injection pressure on smoke opacity of diesel fueled engine

BTE, BSFC, and smoke opacity were measured and calculated at different fuel injection pressure for preheated Jatropha oil. BSFC decreases as the load increases Figure 6. But, at higher loads, BSFC increases. Lowest BSFC (0.31 kg/kWh) was found at 195 bars. Maximum thermal efficiency (30.3%) was found at 195 bar at 7.35 kW rated load Figure 5.

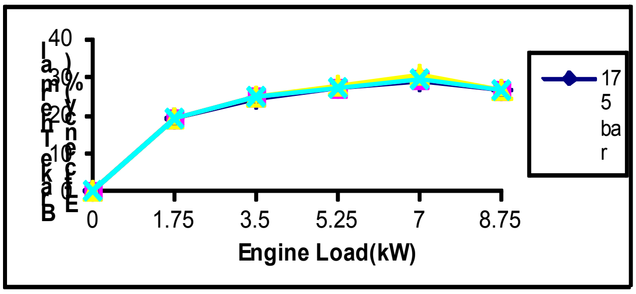

Fig. 5- Effect of fuel injection pressure on brake thermal efficiency of jatropha oil (preheated) fueled engine

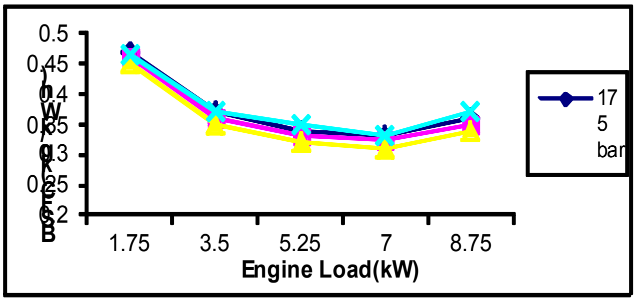

Fig. 6- Effect of fuel injection pressure on brake specific fuel consumption of jatropha oil (preheated) fueled engine

Thermal efficiency decreases when fuel injection pressure either decreases or increases from 195 bar. Smoke opacity was also lowest at 195 bar. Smoke opacity was 33% at 195 bar and at 75% of rated load as shown in Figure 7.

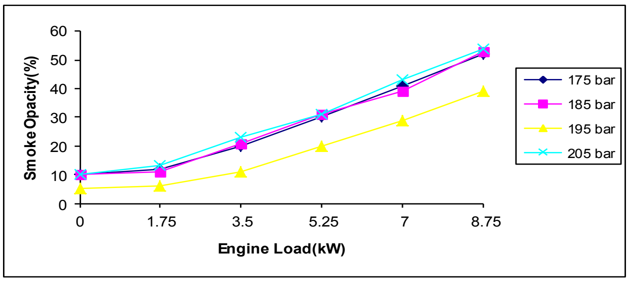

Fig. 7-Effect of fuel injection pressure on smoke opacity of jatropha oil (preheated) fueled engine

Based on BSFC, BTE, and smoke opacity, 195 bar was also found optimum fuel injection pressure for preheated Jatropha oil. Heating the oil reduces the viscosity of Jatropha oil and for pre-heated Jatropha oil also, same optimum fuel injection pressure as that for diesel was found.

Effect of varying fuel inlet temperature on performance and emission of engine

The engine tests were conducted for performance and emission using unheated Jatropha oil and preheated Jatropha oil. The baseline data were generated using diesel. Diesel fuel operation shows lowest BSFC as shown in Figure 9.

Higher BSFC was observed when running the engine with Jatropha oil. Lower calorific value of Jatropha oil leads to increased volumetric fuel consumption in order to maintain similar energy input to the engine. Figure 8 show that the BTE of preheated Jatropha oil was found slightly lower than diesel.

The possible reason may be higher fuel viscosity. Higher fuel viscosity results in poor atomization and larger fuel droplets followed by inadequate mixing of vegetable oil droplets and heated air. However, BTE for preheated Jatropha oil was higher than unheated Jatropha oil. The reason for this behavior may be improved fuel atomization because of reduced fuel viscosity. Figure 10 shows that the smoke opacity for Jatropha oil operation was greater than that of diesel. Heating the Jatropha oil result in lower smoke opacity compared to unheated oil but it is still higher than diesel.

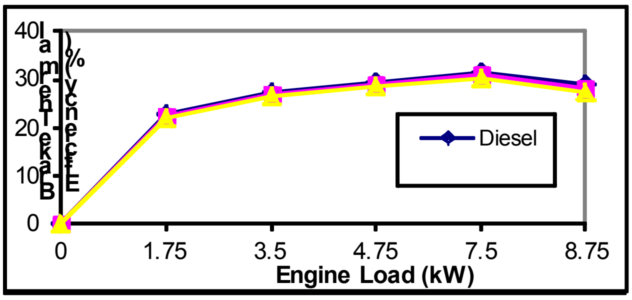

Fig. 8- Brake thermal efficiency of diesel, jatropha oil (preheated) and jatropha oil (unheated) fueled engine

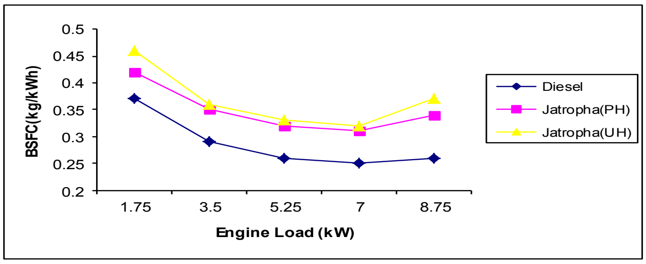

Fig. 9- Brake specific fuel consumption of diesel, jatropha oil (preheated) and jatropha oil (unheated) fueled engine

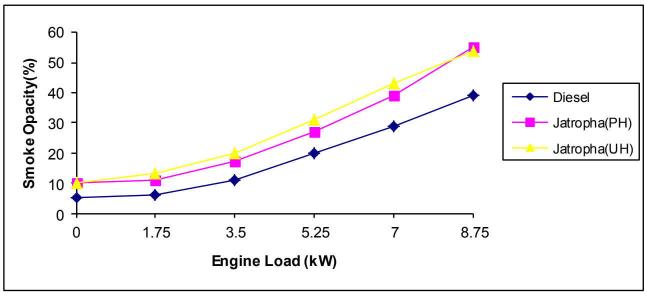

Fig. 10- Smoke opacity of diesel, jatropha oil (preheated) and jatropha oil (unheated) fueled engine

Comparison of theoretical and experimental results

The brake thermal efficiency of compression ignition engine fueled by diesel, jatropha oil (preheated) and jatropha oil (unheated) is compared with that obtained from the theoretical model as shown in Figure 11.

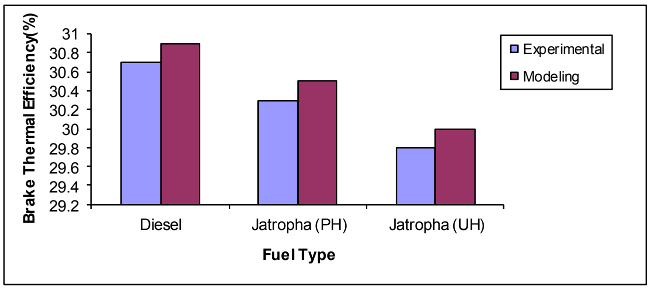

Fig. 11- Comparison of brake thermal efficiency

The brake thermal efficiency of diesel engine is slightly higher with

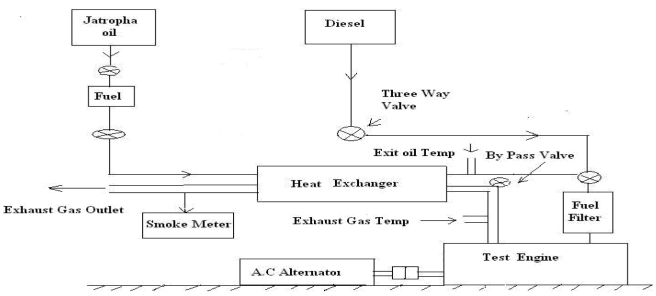

Fig. 12- Schematic diagram of experimental setup

preheated jatropha oil fueled engine. The brake thermal efficiency of the engine decreases with using unheated jatropha oil. The calorific value of jatropha oil is lower than (about 14%) that of diesel because of the presence of oxygen in its molecule. Hence, the brake thermal efficiency of jatropha oil (unheated) fueled engine is lower as compared to that of diesel fueled engine. The variation in experimental and theoretical results may be due to the fact that in theoretical model homogeneous mixture with complete combustion is assumed. But in general, it is difficult to attain complete combustion. Despite the simplification resulting from the assumed hypothesis and empirical relations, the developed simulation proved to be reliable and adequate for the proposed objectives.

Conclusion

A mono zone thermodynamic model is developed for analyzing the performance characteristics of the indirect injection compression ignition engine. The model is developed in such way that it can be used for characterizing any hydrocarbon-fueled engines, as diesel or biodiesel. The modeling results showed that, with increase in fuel injection pressure brake thermal efficiency is increased and brake specific fuel consumption and smoke opacity are decreased. The performance characteristics of the engine follow the same trend for all fuels as diesel, preheated jatropha and unheated jatropha oil. The predicted results are compared with the experimental results of the engine fueled by diesel, jatropha (PH) and jatropha (UH). This model predicted the engine performance characteristics in closer approximation to that of experimental results. Hence, it is concluded that this model can be used for the prediction of the performance characteristics of the compression ignition engine fueled by any type of hydrocarbon fuel.

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Nomenclature

A_n cross sectional area of nozzle, m^2
D cylinder bore, m
L connecting rod length, m
S stroke length, m
m mass of the mixture (air + fuel) contained in the combustion chamber, kgs
N engine speed, rpm
P cylinder pressure, pascal
Q heat energy, joule
R universal gas constant, kJ/molK
Re Reynolds's number
T temperature, K
V instantaneous cylinder volume, m^3
V_p mean piston speed, m/s
V_θ volume at any crank angle, m^3
V_c clearance volume, m^3

Greek

θ crank angle ,degrees
$\Delta\theta$ computational step, °CA

ΔP pressure difference, pascal
ρ density, kg/m^3
v kinematic viscosity, m^2/sec
σ surface tension of fuel, N/m
γ specific heat ratio, dimensionless
μ dynamic viscosity, kg/m s
k thermal conductivity, W/m K

Subscripts
a air
c carbon
com combustion
f fuel
h hydrogen
g gross
inj fuel injection
min minimum
n species number, or nozzle hole
o oxygen
p piston
r released
w wall
0 initial value
1 state at the beginning of time step
2 state at the end of time step

Abbreviations
BTE brake thermal efficiency
BDC bottom dead center
BSFC brake specific fuel consumption (kg/kWh)
EVO exhaust valve opening
EES engineering equation solver
IVC inlet valve closing
rpm revolutions per minute
PH preheated
TDC top dead center
UH unheated

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